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Design of battery operated motorised screw jack

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Abstract

The screw jack is a device for lifting heavy loads by applying comparatively small at its handle. The common problem faced in the conventional screw jack is that it requires great efforts to raise or lower the load. For heavy goods carrier vehicle, this jack is not useful and hence hydraulic jack is preferred which is costly. Today's requirement of jack is that it should require low effort and low cost so the 'Modified Jack' is compromise between hydraulic jack which is costly and conventional screw jack which requires high effort for heavy load. The design of this screw jack to reduce man power, to increase the efficiency, to reduce the work load, to reduce the production cost, less maintenance.

Keywords: screw jack, hydraulic, motorised, gear

1. Introduction

Screw jack consists of threaded rod called as screw rod or power screw. The screw has threads on its outer surface which bits into the internal threads of the nut. The load to be lifted is placed on screw head and the application of effort at the end of handle lifts or lowers the same. A common conventional screw jack is Bottle type jack, Hydraulic jack, Pneumatic jack,

S. No.	Type of jack	Advantages	Disadvantages
1.	Bottle type jack	Cost is less. Light in weight, hence portable	As load increases effort required increases more rapidly. In case of over loading damage occur.
2.	Hydraulic Jack	Lifts heavy load with less effort. Movement from standstill is possible with full force. Compact in construction. Protection from overload damage.	Cost is high. Overall system is bulky. Efficiency is less. Noisy due to shocks in piping
3.	Pneumatic jack	Operation is cleaner than hydraulic jack. Protection from overload damage	Working cost is high due to high compression cost. Actuators are bulky than hydraulic jack.

2. Parts of screw jack

Fig 1: Worm and worm wheel

Fig 2: Screw Rod

Fig 3: Bearing

Fig 4: Base

Fig 5: Motor

3. Working Principle

The jack consist of worm, worm wheel, screw, nut, bearing all housed inside the body of jack. The worm is engaged with worm wheel. Worm shaft is connected to motor shaft by belt-pulley arrangement. As Motor Starts rotating the worm shaft rotates and hence worm rotates, worm wheel rotates and screw comes up. As we change the direction of rotation of motor the screw comes down.

4. Design of parts

4.1 Design of screw

Material Selection Referring to IS designation of carbon steel according to IS:
1570-1979, the material for power screw is 45C8 (0.4-0.5 C, 0.6-0.9Mn) which is plain and medium carbon steel. For which δyt = 360 $\mathrm{N/mm^2}$

Yield strength - For steel screw material with steady load factor of safety is,

F.O.S. =4, δyc = δyt = 360 N/mm

Compressive stress=δc =δyc (from normal stress theory equation.)

$$= \frac{360}{4} = 90\ \mathrm{N/mm^2}$$

Now δys = yield strength in shear, δys = $\frac{1}{3}\partial yt$

$$\frac{\delta ys}{f.o.s} = \frac{1}{\sqrt{3}}\frac{\delta yt}{f.o.s} = \frac{1}{\sqrt{3}}\frac{360}{4} = 51.96\ \mathrm{N/mm^2}$$

An axial load on screw rod produces direct compressive stress which is given by,

$$W=A.\ \delta c = \frac{\pi}{4}(dc)^2.\delta c$$

We take W = 2000Kg = 20000N, W = $\frac{\pi}{4}(dc)^2$.90, Dc= 16.82mm

To account for tension let us provide 20% extra.
Do = dc + 20%= 16.82+ 0.2×16.82=20.18 mm.
Referring basic dimensions of square thread according to IS: 4694-1968 we have

Dc = 21mm, do = 26mm, p = 5mm, dm = $\frac{do+dc}{2} = \frac{21+26}{2} = 23.5\ mm$

4.2 Check for self-locking
For single start screw rod with square threads, Lead =Pitch, 1=p=5mm

For grease lubrication c.o.f for screw nut is taken as µ =0.06 to 0.09 =0.08

Now Helix angle a = $tan^{-1}(\frac{1}{\pi dm}) = tan^{-1}(\frac{1}{\pi * 23.5})$ =3.87°

Friction angle: $tan^{-1}(\mu)\ \emptyset = tan^{-1}(\mu) = tan^{-1}(0.08)$ = 4.57°

As $\emptyset > \mu$ i.e Friction angle is greater than helix angle condition for self-locking
Hence design is safe for slipping.

4.3 Check for stresses induced in screw
Compressive stresses due to axial load.

$\delta c = \frac{W}{A} = \frac{W}{\pi/4 (dc)^2} = \frac{20000}{\pi/4 (21)^2} = 57.74\ N/mm^2$.

Shear stress due to torque is given by,

T= $\frac{16\ T}{\pi\ dc^3}$ But T= P× $\frac{dm}{s}$

= W tan (a+Ø) $\frac{dm}{2}$ = 20000 tan (3.87+4.57) $\frac{23.5}{2}$ = 34869.44 N/mm

T = $\frac{16 * 34869.44}{\pi * (21)^3}$ = 19.18 N/mm^2

Maximum principal stress

δc (max) = $\frac{1}{2}(\delta c + (\delta c^2 + 4T)^{1/2})$ = $\frac{1}{2}(57.54 + (57.54)^2 + 4$ = 63.14N/mm^2 >90N/mm^2

Hence design is safe against compressive stress.
The maximum shear stress is given by,

T (max) = $1/_2(\delta c^2 + 4T^2)^{1/2}$ =$1/_2(57.54)^2 + 4(19.17)^2)^{1/2}$=34.58$N/mm^2 < 51.96 N/mm^2$

Hence design is safe against shear stress

4.4 Check against Bulking
Maximum lift of screw =150mm.
Effective lift of screw for buckling.
Lift of screw = 1+$1/_2$ Height of nut. = 150+$1/_2$ ×70 =185mm.

When the screw reaches the maximum lift it can be regarded as a strut whose lower end is fixed and load end is free.

Critical load, Wσ =Acσyt ($\frac{1 - 6yt\,(L/_K)^2}{4C\,\pi^2 E}$)

For one end is fixed and other end is free, C=0.25
Also, K+0.25 dc = 0.25×21 = 5.25 mm.

Wcr = $\frac{\pi}{4}(21) \times 2 \times \frac{1 - 360(185/_{5.25})^2)}{4 \times 0.25 \times \pi^2 \times 2.1 \times 106}$ =122000.5N so as F.O.S=6.1>4…..our design is safe.

4.5 Design of NUT
To minimize friction, material used for nut is weaker than screw material. So material of nut is cast iron.

Bearing pressure =120 Kg/Cm^2 =12 N/mm^2

Pb = $\frac{W}{\pi * dm * t * n}$ = $\frac{20000}{\pi * 23.5 * t * n}$

n =9.03 =10
Number of thread in contact with screwed spindle=10
Thus Height of nut =H=n*p=10*5=50mm.

4.6 Checking for shear stress in nut

σs (act) = $\frac{W}{\pi n d2t} = \frac{20000}{\pi * 14 * 50 * 10}$ = 0.91N/mm^2 < 23 N/mm^2

Hence design is very much safe.

Thickness of thread $P/_2 = 5/_2 = 2.5$ mm.

4.7 Design of worm gear
Torque transmitted by screw T=T1+T2
Where; T1=34869.44 N/mm = 3486.944 Kgf/mm., T2= $\frac{W.dred.\mu}{2}$

dred = $\frac{2(D2^3 - D1^3)}{3(D2^2 - D1^2)} = \frac{2(50^3 - 40^3)}{3(50^2 - 40^2)}$ = 45.18 = 45mm.

T2 = $\frac{20000 * 45 * 0.08}{2}$ = 36000 N.mm = 3600 Kgf.mm
T=T1+T2= 3486.9+3600 = 7086.9 Kgf.mm
Now, consider approximate efficiency in worm gear drive.
For Z=1 n=0.70 to0.73 T=7086.9*0.73 =5173.43 kgf.mm
T=T*Ka*K K= Load concentration factor =1, when load is constant. Kd =1 for V2 =3 m/s.
[T] =5173.43*1*1 =5173.43 Kgf.mm

4.8 Designation of worm pair
Z4/Z2/q/m=2/30/10/4
Now check for bending strength:

Fb = $\frac{1.9[Mt]}{M^3 * q * Z2 * Yv}$

Hence for virtual no. of teeth.
Y = $\tan^{-1}(Z4/9) = \tan^{-1}(2/10)$ = 11.30°
Zv= $Z2/\cos^3 y = 30/\cos^3(11.30°)$ = 31.81=35
For 35teeth Yv=0.452

Fb= $\frac{19[Mt]}{Mx^3 * q * Z2 * Yv} = \frac{19[Mt]}{0.4^3 * 10 * 30 * 0.452}$=113.26 kgf/$cm^2$ < 300 kgf/cm^2

Hence design is safe for bending.

4.9 Basic dimension of worm gear pair
Center distance=80mm Face width of wheel b=0.75
D1 reference dia=q Mx=10×4=40 mm.
Face width of worm wheel b=0.75d1=0.75×40=40 mm.
Axial module Mx=4 mm, No.of starts Z4= 2
Z2=I ×Z4=15×2=30. Diameter factor= 10

4.10 Designation of worm wheel

Z1/Z2/q/m=2/30/10/4.

Axial pitch =px =$\pi \times$ mx= $\pi \times 4$=12.56 mm

4.11 Lead angle on reference cylinder

$\propto$ = 11 18' 36"

Bottom clearance = c= 0.2 to 0.3 mx= 0.25 mx= 1 mm

Addendam $modi^n$ coeff. = x = [a/mx] – 0.5[q +z2] = [80 /4] – 0.5[10+30] = 0

4.12 Worm

Reference dia = d1 = q × mx = 10×4= 40 mm
Tip dia da1 = d1 + 2fo mx= 40 + 2 ×1× 4= 48 mm
Root dia da1 =df1 = d1 – 2fo mx – 2c = 40- 2× 1×4- 2×1= 30 mm
Pitch dia d= mx [q + 2x] = 4 [10+ 2×0] = 40 mm
Length of worm = L > [11+0.06 Z 2] mx ie ≥ 52 mm [p.s.g table]

4.13 Worm Wheel [In axial section]

Reference dia = d2 =Z2 mx = 30× 4= 120 mm
Tip dia = dq2 = [Z2 + 2 fo + 2x] mx = [30 +2 × 1+ 2× 0] 4= 128 mm
Root dia =df2 = [Z2 -2 fo] mx – 2c= [30-2 × 1] 4-2×1 = [30-2 × 1] 4-2×1= 110 mm
External dia of worm wheel
De2 = dq2 + mx = 128 + 4= 132 mm
Pitch dia d2 =d2', d2 = 120 mm

4.14 Worm wheel Throat

Throt tip radius R1 = d1/2- fo. mx = 40/2 - 1× 4= 16 mm
Throt root radius R2 = d1/2+fo mx + c= 40/2 +1×4 +1= 25 mm
Tip relief radius = r1= 0.1 mx=0.1×4= 0.8 mm
Nominal tooth thickness on reference dia

Axial section s= $\frac{\pi mx}{2} = \frac{\pi \times 4}{2}$ = 6.28 mm

Nominal Tooth thickness on reference dia in normal section

Sa= $\frac{\pi}{2} \times mx \times \cos y = \frac{\pi}{2} \times 4 \times \cos(11.30)$ = 6.16 mm

So friction angle = tan -1µ
For rubbing /Sliding velocity = 0.25m/sec, µ=0.8- friction coeff. For velocity = 0.25m/sec
Friction angle +tan -1[0.08]

$\mu = \frac{\tan y}{\tan(y+\mu)} = \frac{\tan(11.30)^{\circ}}{\tan(11.30+4.57)}$=0.70 µ=70 %

4.15 Worm

d= pitch dia, d1 = reference dia, df2= Root dia. da1=tip dia

4.16 Worm Wheel

b= face width, d2=Pitch dia, df2=Root dia, da2=Tip dia, de2=External dia

4.17 Worm Wheel Throat

r1=Tip relief radius, r2=Root relief radius, R1=Tip throt radius, R2=Root throt radius

4.18: Design Of bearing

Deep Groove ball bearing

Life of bearing in million revolution L= $\frac{60LHn}{10^6}$ Where, LH = bearing life in hour
n = speed of rotation in RPM as the rotation of shaft is manual thus speed is 60 Rpm assume expected life 4000-8000 hr

We choose expected life as 8000 hr life of bearing, L= $\frac{60LHn}{10^6}$ = $\frac{60\times 8000\times 60}{10^6}$ = 28.8 hr

Probabality of survival

$\frac{L}{L90}$=log $(\frac{I}{R})$ × log 90$(\frac{1}{R90})$ $\frac{1}{b}$

Where, L=Required life of bearing in mr, L90= calculated life of selected bearing for givan load for 90% survival
b=1.34 ……radial load
For deep Groove ball bearing 28.8=log $(\frac{1}{0.92})$ 1/1.34, L90= log $(\frac{1}{0.92})$ 1/1.34, L90= 36.30mr

For selection of bearing we will consider radial load which is coming on cylinder roller bearing

L90 = [$\frac{C\ req}{fr}$] k

Where, Fr = radial force acting on worm, K =10/3 for roller bearing, 28.9 =ft tan ∝
Where, ∝= Normal pressure angle= 20, ft = 862.21 N, fr= 862.21×tan 20°, = 313.8 N
C req =fr [L90] k= 861.8, N Root Dia of worm =30 mm
We choose the bearing of C = 2700 N
D = outer dia of bearing, d= inner di of bearing, Bearing of basic design no = 6004
D= 42 mm, d=20 mm, B=12mm
Calculate equivalent dynamic load from the equation,
P= (Xfr+Yfa) s
Where, P=equivalent load, s = service factor, Fa = Axial load, X=Radial factor, Y= Axial factor
Let, Fa = 0, X = Y=1
P= (1×313.8+0) 1.3=407.94, P=408N

L= [$\frac{C}{P}$] K

K=10/3………..For roller bearing

L= [$\frac{7300}{408}$] 3.33

L=14871.5 Mr>28.8 Hr
This life is very much greter than so bearing used will work for long period.
Thrust ball bearing:
The thrust ball bearing is selected for dia. Of shaft =26mm
d=inner race diameter =30mm, D=outer race diameter=52mm, H=height of bearing=16mm
Design of Motor

P = $\frac{2\pi NT}{60}$ the maximum load is assumed =2000 kg
F = 2000×9.81 =19620
T = F× perfendicular distance=19620×7.5 =147150 Nmm, T= 147.15 Nm

P = $\frac{2\pi NT}{60}$ = $\frac{2\pi\times 60\times 147.15}{60}$ = 924.5 watt

1HP = 746 Watt = $\frac{924.5}{746}$ =1.23 Hp

Therefore we selected Power of motor (1.23) at (60 rpm) (AC motor)

5. Conclusion

Screw Jacks are the ideal product to push, pull, lift, lower and position loads of anything from a couple of kilograms to hundreds of tonnes. The need has long existed for an improved portable jack for automotive vehicles. It is highly desirable that a jack become available that can be operated alternatively from inside the vehicle or from a location of safety off the road on which the vehicle is located. Such a jack should desirably be light enough and be compact enough so that it can be stored in an automobile trunk, can be lifted up and carried by most adults to its position of use, and yet be capable of lifting a wheel of a 4,000-5,000

pound vehicle off the ground. Further, it should be stable and easily controllable by a switch so that jacking can be done from a position of safety. It should be easily movable either to a position underneath the axle of the vehicle or some other reinforced support surface designed to be engaged by a jack.
Thus, the product has been developed considering all the above requirements. This particular design of the motorized screw jack will prove to be beneficial in lifting and lowering of loads.

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